

Optimum absorber temperature of a once-reflecting full conical concentrator of a low temperature differential Stirling engine

Bancha Kongtragool, Somchai Wongwises*

*Fluid Mechanics, Thermal Engineering and Multiphase Flow Research Laboratory (FUTURE),
Department of Mechanical Engineering, King Mongkut's University of Technology,
Thonburi, Bangmod, Bangkok 10140, Thailand*

Received 13 August 2004; accepted 12 January 2005

Available online 25 February 2005

Abstract

This paper provides a theoretical investigation on the optimum absorber temperature of a once-reflecting full conical concentrator for maximizing overall efficiency of a solar-powered low temperature differential Stirling engine. A mathematical model for the overall efficiency of the solar-powered Stirling engine is developed. The optimum absorber temperature for maximum overall efficiency for both limiting conditions of maximum possible engine efficiency and maximum possible engine power output is determined. The results indicated that the optimum absorber temperatures calculated from these two limiting cases are not significantly different. For a given concentrated solar intensity, the maximum overall efficiency characterized by the condition of maximum possible engine power output is very close to that of the real engine of 55% Carnot efficiency, approximately.

© 2005 Published by Elsevier Ltd.

Keywords: Stirling engine; Solar-powered heat engine; Conical concentrator

* Corresponding author. Tel.: +662 4709115; fax: +662 4709111.

E-mail address: somchai.won@kmutt.ac.th (S. Wongwises).

Nomenclature

A_O	cone opening area in m^2
A_H	absorber area or heater convection heat transfer area (m^2)
C	concentration factor
C_1, C_2	the factors representing the performance of the two ways of energy collection of the conical concentrator, defined by Eqs. (5), (6), (12) and (13)
E	overall efficiency
E_C	concentrator efficiency
E_E	engine efficiency
E_{IT}	indicated thermal efficiency
E_{Carnot}	complete reversible Carnot efficiency
$E_{Curzon-Ahlborn}$	Curzon–Ahlborn efficiency
$E_{Endo-reversible}$	endo-reversible Carnot-like engine efficiency
$F\phi$	incidence angle reduced factor
h_H	heater convection heat transfer coefficient ($W/m^2 K$)
I	direct solar flux intensity (W/m^2)
K_S	Stirling coefficient = E_{IT}/E_{Carnot}
K_1	a constant, defined by Eq. (30)
K_2	a constant, defined by Eq. (31)
q	total energy received by the absorber plate (W)
q_{CH}	convection loss (W)
q_{in}	useful energy (W)
q_{RH}	radiation loss (W)
q_S	total solar energy input to the concentrator (W)
R	radius of the cone opening (m)
R	radius of the absorber plate (m)
T_A	ambient temperature (K)
T_C	engine cooler temperature (K)
T_H	absorber or engine heater temperature (K)
$*T_H$	optimum absorber temperature (K)
T_{Sky}	sky temperature (K)
T_1	the cold-side working fluid temperature (K)
T_3	hot-side working fluid temperature (K)

Greek letters

α	absorptivity of absorber plate
ε	emissivity of the absorber plate
ϕ	cone included angle in degrees, used in Eq. (7)
ρ	reflectivity of reflector
σ	Stefan–Boltzmann constant, $5.667 \times 10^{-8} W/(m^2 K^4)$
τ	transmissivity of cover plate

1. Introduction

Solar energy is an attractive renewable energy source that can be used as an input energy source for heat engines. In fact, any heat energy source can be used in a Stirling engine. The solar radiation can be focused onto the displacer hot-end of the Stirling engine, thereby creating a solar-powered prime mover. The direct conversion of solar power into mechanical power reduces both the cost and complexity of the prime mover.

When a solar collector is used as a heat input source of a heat engine for power generation, one of the design objectives is to optimize the overall system performance. In general, the collector works best at low temperatures and its efficiency decreases with increasing temperature. However, the heat engine is most efficient with heat input at high temperatures and its efficiency increases with increasing temperature.

The overall efficiency of the direct solar-powered heat engine is the product of the solar collector efficiency and the heat engine efficiency. The dependence of the collector, engine, and system efficiency on temperature is shown in Fig. 1. This contrary temperature-efficiency relationship indicates that any solar-powered heat engine will have an optimum absorber temperature. Therefore, in solar-powered heat engine design there is a need to know the optimum absorber temperature in order to operate the system at its maximum overall efficiency.

The optimum solar collector temperature has been studied by many researchers. Many works on low temperature differential (LTD) Stirling engines and solar-powered Stirling engines including technology and optimization have been investigated in the authors' former works [1–3]. Among many researches, some closely related works are as follows:

In 1977, Howell and Bannerot [4] determined the optimum value of the outlet temperature of the solar collector to maximize the work output of idealized Carnot, Stirling, Ericsson, and Brayton engines powered by a solar collector. Their mathematical model is formulated by using a ratio of useful energy provided by the collector to

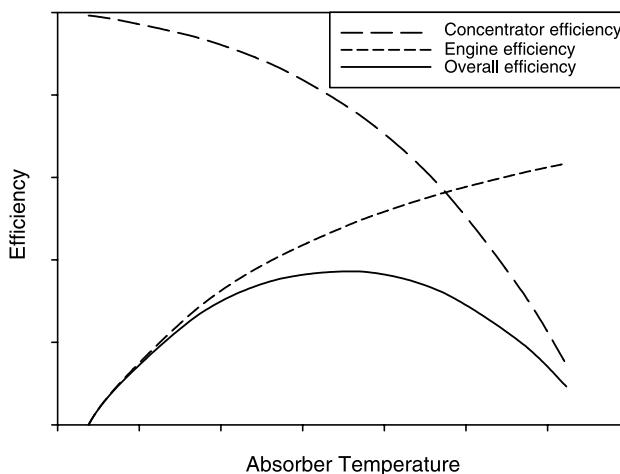


Fig. 1. Typical temperature-efficiency relationship of solar-power heat engine.

the absorbed solar energy and a ratio of the useful work to the absorbed solar energy. Heat losses from collector were both the radiation and convection heat loss. For Stirling engine efficiency, the formula for an imperfect regeneration Stirling cycle was used. However, in their analysis the hot-side working fluid temperature was the heater temperature and the cold-side working fluid temperature was the cooler temperature approximated as an ambient temperature.

In 1988, Gordon [5] examined the accuracy of the energy optimization of solar-driven heat engines. The results were obtained for the two limiting cases of maximum efficiency and maximum power. The effect of radiation and convection heat loss was studied separately; the Carnot and Curzon–Ahlborn efficiencies were used in the analysis.

In 1993, Eldighidy [6] used the same concept as Howell and Bannerot [4] to theoretically investigate the optimum outlet temperatures of the solar collector for maximum work output for an ideal regeneration air-standard Otto cycle. The effect of radiation and convection loss from the collector on the optimum outlet temperature was shown. Idealized air-standard cycle efficiency was used in analysis. However, in his analysis the temperature difference between the absorber and the hot-side working fluid of 5 K was assumed.

In 1996, Chen, Sun and Wu [7] presented the optimum collector temperature for solar-powered heat engines. Four conditions of collector heat loss are modeled with Carnot and Curzon–Ahlborn efficiencies. However, the application on their models was not shown in the paper.

Although many researchers have studied the optimum solar collector temperature, there still remains room for study on Stirling engines operating at low temperatures. LTD Stirling engine powered by a once-reflecting full conical concentrator is one that has received little attention in literature and should be studied in detail.

This article is a theoretical investigation on the optimum absorber temperature required for a LTD Stirling engine to operate at its maximum overall efficiency. The aim of this article is to provide the basis for the design of a real solar-powered LTD Stirling engine operated with a low cost concentrator.

2. Mathematical model

A schematic diagram of a solar-powered Stirling engine is shown in Fig. 2. The analysis of the problem includes the mathematical models for the solar concentrator, Stirling engine, and the combination of solar collector and Stirling engine. These models are described as follows:

2.1. Solar collector

In principle, both the concentrating and non-concentrating solar collector can be used to power the Stirling engine. Since the practical temperature limit of the flat-plate collector is around 100 °C above the ambient temperature [7], a more efficient Stirling needs a concentrating collector [8].

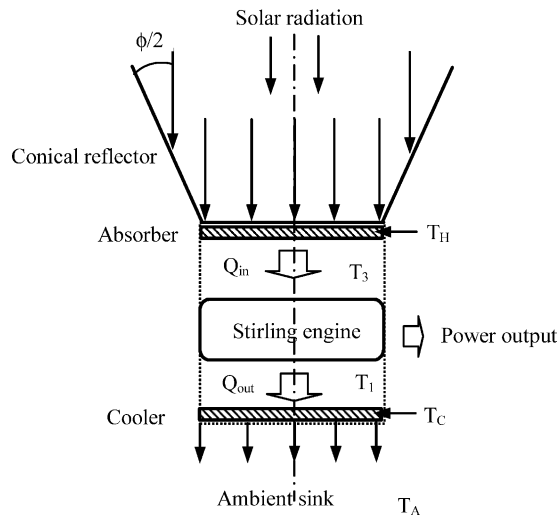


Fig. 2. Schematic diagram of a solar-powered Stirling engine.

The concentrator considered is limited to a reflector, called the once-reflecting cone, which can be characterized in that every sunray entering parallel to the axis and striking the cone is reflected directly on the absorber plate only. This configuration reduces the cone length, minimizes reflection losses, and is most allowing for focusing errors [9]. The following analysis is applied to the condition that the cone axis is always parallel to the sunrays. Once-reflecting full conical reflector is a cone where, [9]:

$$R = r(1 + 2 \cos \phi) \quad (1)$$

where R is radius of the cone opening in m, r is radius of the absorber plate in m, and ϕ is cone included angle in degrees.

The total solar energy input to the concentrator is given by [10]:

$$q_s = I A_O \quad (2)$$

where I is direct solar flux intensity in W/m^2 , and A_O is cone opening area in m^2 .

For a sunray entering parallel to the cone axis, the solar power concentrated onto the absorber plate with a transparent cover is given by [9]:

Concentrated solar power, q = Direct radiation onto absorber + Reflected radiation by cone onto absorber

$$q = I A_H C_1 + I (A_O - A_H) C_2 \quad (3)$$

$$q = C_1 I \pi r^2 + C_2 I \pi (R^2 - r^2) \quad (4)$$

where q is total power absorbed by the absorber plate in W, A_H is absorber area in m^2 , C_1 is the factor representing direct radiation onto absorber, and C_2 is the factor representing reflected radiation by cone onto absorber.

The factor C_1 depends on the transmissivity of the cover and the absorptivity of the absorber plate. The factor C_2 is product between the reflectivity of the reflector, incidence angle reduced factor, and the combined transmissivity and absorptivity of the absorber.

$$C_1 = \alpha\tau \quad (5)$$

$$C_2 = \alpha\tau\rho(F\phi) \quad (6)$$

where α is absorptivity of absorber plate, τ is transmissivity of absorber plate, ρ is reflectivity of reflector and $F\phi$ is incidence angle reduced factor [9,11]:

$$F\phi = -1.06332 \times 10^{-9}\phi^3 + 0.396 \times 10^{-6}\phi^2 - 48.86922 \times 10^{-6}\phi + 0.9217 \quad (7)$$

Eq. (4) can be written as:

$$q = I\pi r^2[C_1 + 4C_2 \cos \phi(1 + \cos \phi)] \quad (8)$$

The concentrating factor, C , is defined as:

$$C = q/I\pi r^2 = [C_1 + 4C_2 \cos \phi(1 + \cos \phi)] \quad (9)$$

Then

$$q = I\pi r^2 C = IA_H C \quad (10)$$

For ideal cone, $C_1 = C_2 = 1$, Eq. (9) give the interesting fact that the concentration factor has a limiting value of $C=9$ as the cone vertex angle approaches zero, as shown in Fig. 3. As the concentration factor approaches 9, cone length increases without bound. However, ideal concentration factors of up to about 7 can be obtained with cones of practical length to diameter ratio [9]. For ideal cone the concentration factor can be determined from:

$$C = A_O/A_H = (1 + 2 \cos \phi)^2 \quad (11)$$

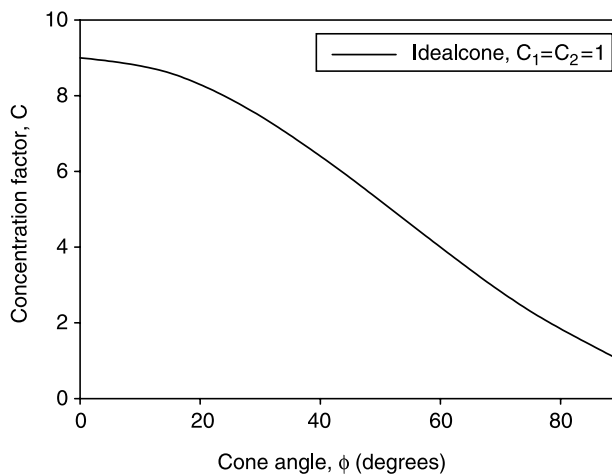


Fig. 3. Ideal concentrating factor of once-reflecting full conical reflector, calculated from Eq. (9).

The following analysis is applied to an opaque absorber. For an opaque absorber, the factor C_1 and C_2 can be determined from:

$$C_1 = \alpha \quad (12)$$

$$C_2 = \alpha \rho (F\phi) \quad (13)$$

Then, the previous equations for an absorber with a transparent cover can be applied to an opaque absorber by using the appropriate factor C_1 and C_2 .

In general, disregard the heat transmission through the cone wall, the energy balance of an absorber plate is [4,7,10]:

Useful energy collected, q_{in} = Concentrated solar energy, q – Radiation loss, q_{RH} – convection loss, q_{CH}

Suppose that the absorber plate operates at a temperature of T_H . Howell and Bannerot [4] and Brinkworth [8] proposed the radiation loss in a simple form:

$$q_{RH} = \varepsilon \sigma A_H T_H^4 \quad (14)$$

where σ is the Stefan–Boltzmann constant, $5.667 \times 10^{-8} \text{ W/m}^2 \text{ K}^4$, ε is the emissivity of the absorber plate. Eldighidy [6] presented the radiation loss in a more complicated form

$$q_{RH} = \varepsilon \sigma A_H (T_H^4 - T_{sky}^4) \quad (15)$$

where [6,12]

$$T_{sky} = 0.0552 T_A^{1.5} \quad (16)$$

where T_A is the ambient temperature, therefore, Eq. (15) becomes:

$$q_{RH} = \varepsilon \sigma A_H (T_H^4 - 9.2845 \times 10^{-6} T_A^6) \quad (17)$$

It should be noted that, if $T_A = 35^\circ \text{C} = 308 \text{ K}$, $T_{sky} = 298 \text{ K} = 0.97 T_A$. However, Kreith and Kreider [10] and Chen et al. [7] proposed the radiation loss in a simpler form:

$$q_{RH} = \varepsilon \sigma A_H (T_H^4 - T_A^4) \quad (18)$$

Several researchers [4,6,12] have proposed the convection loss in the form

$$q_{CH} = h_H A_H (T_H - T_A) \quad (19)$$

where h_H is the heater convection heat transfer coefficient and A_H is the heater convection heat transfer area. Brinkworth [8] suggested the value of $h_H = 4 \text{ W/m}^2 \text{ K}$ for still air and $h_H = 30 \text{ W/m}^2 \text{ K}$ for the wind velocity of 10 m/s.

If Eqs. (18) and (19) are used to represent the radiation and convection loss, then, the useful energy collected in a general case of absorber heat loss is:

$$q_{in} = I A_H C - \varepsilon \sigma A_H (T_H^4 - T_A^4) - h_H A_H (T_H - T_A) \quad (20)$$

In the case of a collector with a high concentration factor, many researchers have made analyses by considering the main part of energy losses to the environment heat sink to occur only by radiation [7,8,10].

The concentrator efficiency is the ratio of the useful collected energy to the solar energy input:

$$E_C = q_{in}/q_s \quad (21)$$

Substitute Eqs. (2) and (20) in Eq. (21):

$$E_C = [IA_H C - \varepsilon \sigma A_H (T_H^4 - T_A^4) - h_H A_H (T_H - T_A)] / IA_O \quad (22)$$

For ideal cone, $A_O = A_H C$, then the concentrator efficiency can be determined from:

$$E_C = 1 - [\varepsilon \sigma / (IC)] (T_H^4 - T_A^4) - [h_H / (IC)] (T_H - T_A) \quad (23)$$

2.2. Stirling engine

The Stirling engine in general is well suited to the solar applications [8,13]. The most critical aspect of a solar-powered engine is the design of the solar collector for the engine. If the absorber (that acts as the displacer cylinder head) cannot effectively give the required heat input then the desired engine power output and efficiency will not be obtained.

Heat engines are usually designed to operate at a point between two limiting cases of practical interest, the maximum possible efficiency and the maximum possible power output. The first limiting case is the Carnot efficiency of the complete reversible heat engine, which represents the maximum engine efficiency. The second limiting case is the Curzon–Ahlborn efficiency of the endo-reversible heat engine that represents the efficiency at the operation of the maximum power output.

The complete reversible Carnot efficiency is given by

$$E_{\text{Carnot}} = 1 - T_C/T_H \quad (24)$$

where T_C is the engine cooler temperature in K (see Fig. 4). To reach the complete reversible Carnot efficiency, the isothermal heating and cooling processes must be infinitely slow enough to ensure that the thermal equilibrium between working fluid and its

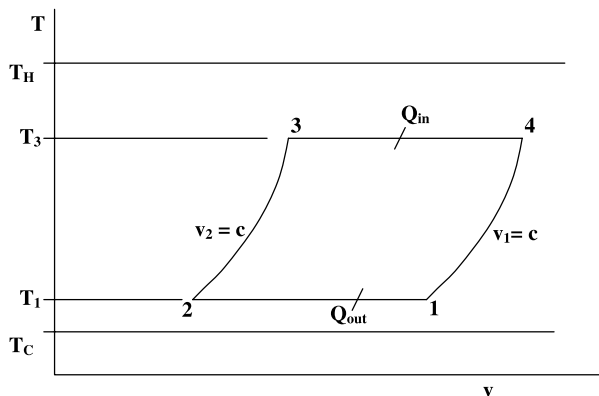


Fig. 4. T-s diagram for Stirling cycle.

heat source or heat sink occurred. Since an infinite period of time is required to get a finite amount of work, the engine power output then approaches zero.

In the endo-reversible heat engine, the two heat transfer processes from the heat source and to the heat sink are considered to be the only irreversible processes in the cycle [7]. In Carnot-like engines the heat exchange between the working fluid and heat source and heat sink is isothermal or near-isothermal [5]. The Stirling engine is a Carnot-like engine. The endo-reversible Carnot-like engine efficiency is lower than the complete reversible Carnot engine efficiency, however it produces useful power output. The endo-reversible Carnot-like engine efficiency is given by [7]

$$E_{\text{Endo-reversible}} = 1 - T_1/T_3 \quad (25)$$

where T_1 is the cold-side working fluid temperature in K, T_3 is the hot-side working fluid temperature in K. Curzon and Ahlborn [14] were the first to derive the endo-reversible Carnot-like engine efficiency under the condition of maximum power output. The Curzon–Ahlborn efficiency is given by [14]:

$$E_{\text{Curzon - Ahlborn}} = 1 - (T_C/T_H)^{0.5} \quad (26)$$

Curzon–Ahlborn efficiency does not represent an upper limit on real heat engine efficiency; however, it only represents an upper limit on real heat engine efficiency under the condition of maximum power output [5].

For a real Stirling engine, the engine efficiency expression can be found in a simple form from the Malmo formula [15]

$$E_{\text{IT}} = K_S(1 - T_C/T_H) \quad (27)$$

where K_S is the Stirling coefficient, the ratio of indicated thermal efficiency to Carnot efficiency. K_S is the proportion of the ideal Stirling cycle efficiency that can be obtained with the present technology [15]. The value of K_S will be found in the range of 0.55–0.88 ($K_S = 0.55\text{--}0.88$) [15]. For well optimized hydrogen engines operated at their maximum efficiency points, K_S is 0.65–0.75, and under special conditions, $K_S = 0.8$ [16].

Other works related to K_S are scarce. However, Brinkworth [8] analyzed the overall efficiency of a solar-powered Stirling engine by using an engine efficiency of 50% of the Carnot efficiency. Likewise, for Stirling engine design, Walker [17] suggested an engine efficiency of 50% of the Carnot efficiency.

2.3. Solar-powered stirling engine

For a solar-powered Stirling engine system consisting of a once-reflecting full conical concentrator and a Stirling engine, the overall efficiency is given by:

$$E = E_C E_E \quad (28)$$

Rearrange Eq. (23) as

$$E_C = 1 - K_1(T_H^4 - T_A^4) - K_2(T_H - T_A) \quad (29)$$

where

$$K_1 = [\varepsilon\sigma/(IC)] \quad (30)$$

$$K_2 = [h_H/(IC)] \quad (31)$$

Then, Eq. (28) becomes:

$$E = [1 - K_1(T_H^4 - T_A^4) - K_2(T_H - T_A)]E_E \quad (32)$$

The limiting expressions for the two limits of maximum possible efficiency and maximum possible power output can be formulated by substituting Eqs. (24) and (26) into Eq. (32). Assuming that the engine design operating temperature falls between these two limits, the optimal operating temperature should also lie between the optimal temperatures determined from these two limiting cases.

It should be noted that the Carnot efficiency is significantly greater than the real engine efficiency; therefore, the overall efficiency calculated with the Carnot efficiency is also much greater than that of the real engine [5].

2.3.1. The condition of maximum possible efficiency

Substitution of Eq. (29) into Eq. (32) gives

$$E = E_C E_{\text{Carnot}} = [1 - K_1(T_H^4 - T_A^4) - K_2(T_H - T_A)][1 - T_C/T_H] \quad (33)$$

To maximizing the overall efficiency, take the derivative of the overall efficiency with respect to the absorber temperature and equate it to zero, $dE/dT_H=0$, the optimum absorber temperature, $*T_H$, for this condition can be obtained from:

$$*T_H^5 - 0.75T_C *T_H^4 + [0.25(K_2/K_1)] *T_H^2 - 0.25[T_A^4 + (K_2/K_1)T_A + 1/K_1]T_C = 0 \quad (34)$$

2.3.2. The condition of maximum possible power output

Substitution of Eq. (26) into Eq. (32) gives:

$$E = E_C E_{\text{Curzon-Ahlborn}} = [1 - K_1(T_H^4 - T_A^4) - K_2(T_H - T_A)][1 - (T_C/T_H)^{0.5}] \quad (35)$$

In the same manner, the optimum absorber temperature for the second condition can be found as:

$$*T_H^{9/2} - 0.875T_C^{0.5} *T_H^4 + 0.25(K_2/K_1) *T_H^{3/2} - 0.125(K_2/K_1)T_C^{0.5} \quad (36)$$

$$*T_H - 0.125[T_A^4 + (K_2/K_1)T_A + 1/K_1]T_C^{0.5} = 0$$

2.3.3. Real Engine

Substitution of Eq. (27) into Eq. (32) gives:

$$E = E_C E_E = [1 - K_1(T_H^4 - T_A^4) - K_2(T_H - T_A)][K_S(1 - T_C/T_H)] \quad (37)$$

It can be seen that Eq. (37) is the same as Eq. (33), except for the constant K_S . In this case, therefore, the same optimum absorber temperature as the case of maximum possible efficiency will be achieved.

2.4. Solution method

Some typical values for the emissivity ε_H , the convection heat transfer coefficient h_H , the ideal concentration factor C , and the solar flux intensity I , are needed in the calculations. The cooler temperature T_C and the ambient temperature T_A are also needed as input data in this calculation. The steps in calculation are as follows:

- Step 1 *Calculate the optimum absorber temperatures:* The optimum absorber temperatures for maximizing overall efficiency at various concentrated solar intensity, IC , can be determined from Eqs. (34) and (36).
- Step 2 *Calculate the concentrator efficiencies:* The concentrator efficiencies at maximum overall efficiency calculated from Eq. (23) by using the optimum absorber temperatures obtained from step 1.
- Step 3 *Calculate the engine efficiencies:* Engine efficiency at maximum possible power output operation is calculated from Eq. (26) with the optimum absorber temperature calculated from Eq. (36). The real engine efficiency is calculated from Eq. (27) by using $K_S = 0.5$ with the optimum absorber temperature calculated from Eq. (34).
- Step 4 *Calculate the overall efficiencies:* The overall efficiencies are the products of the concentrator efficiencies obtained in step 2 and the engine efficiency obtained in step 3.

3. Results and discussion

The results presented are based on some typical values for radiation and convection parameters; $\varepsilon = 0.39$, $\sigma = 5.667 \times 10^{-8} \text{ W/m}^2 \text{ K}^4$, and $h_H = 4 \text{ W/m}^2 \text{ K}$. The ideal concentration factor is fixed at the practical limit of $C = 7$. The solar flux intensities in the range of 100–1000 W/m^2 , with increments in steps of 100 W/m^2 , are used in the calculations. All calculations assumed an absorber area of 1 m^2 and an ambient temperature of $T_A = 35^\circ \text{C}$. The cooler temperatures of $T_C = 45, 55, \text{ and } 65^\circ \text{C}$ are used in this study.

The optimum absorber temperatures for maximizing overall efficiency at various concentrated solar intensity, IC , are shown in Figs. 5 and 6, respectively. It can be seen that the optimum absorber temperature for both limiting cases increases with increasing concentrated solar intensity and cooler temperature.

As shown in Fig. 7, the optimum absorber temperature in the condition of maximum possible power output is slightly greater than the case of maximum possible efficiency. The difference in optimum absorber temperature for these two limiting cases also increases with an increasing concentrated solar intensity. However, this difference in optimum absorber temperature is not significance since its value is only a few percentages.

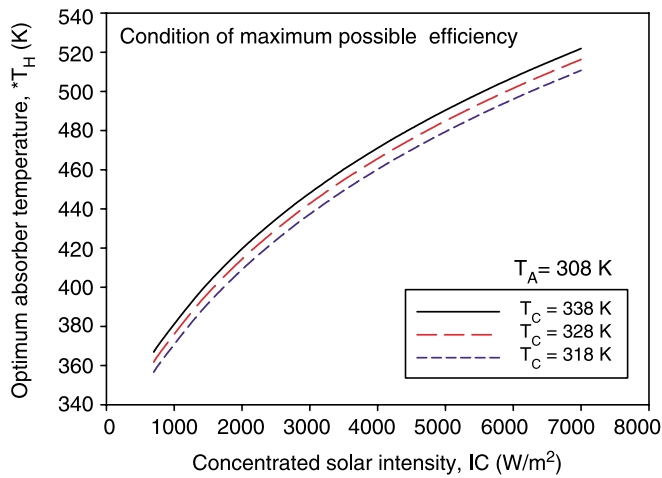


Fig. 5. Optimum absorber temperature against concentrated solar intensity in case of maximum possible efficiency (and also real engine).

Therefore, either the condition of maximum possible efficiency or the condition of maximum possible power output can serve as a basis for calculating the optimum absorber temperature.

For a given cooler temperature, the optimum absorber temperature with the concentrator, engine and overall efficiency can be plotted against the concentrated solar intensity as shown in Fig. 8. To clarify examine the effects of the cooler temperature on these efficiencies; these efficiencies at various cooler temperatures are separately plotted against the concentrated solar intensity.

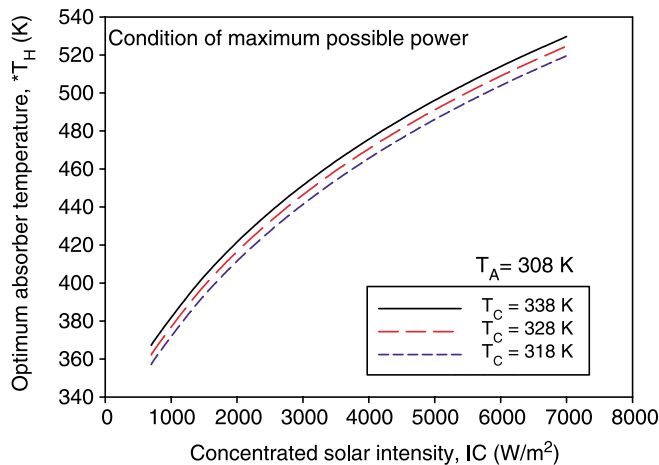


Fig. 6. Optimum absorber temperature against concentrated solar intensity in case of maximum possible power output.

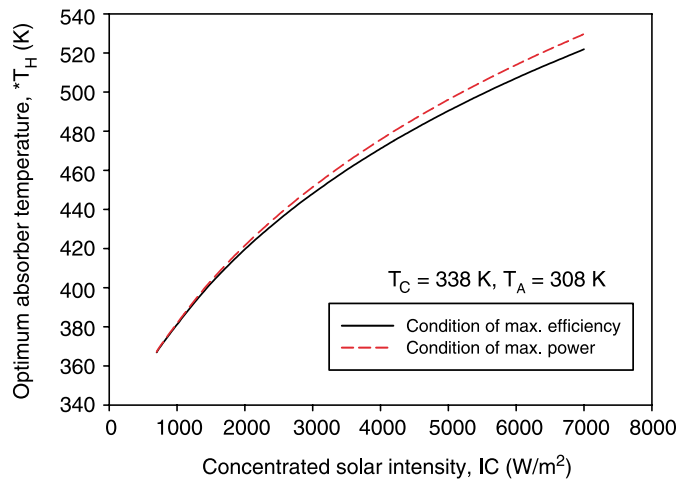


Fig. 7. Optimum absorber temperature against concentrated solar intensity, comparison between two limiting cases.

The concentrator efficiencies at maximum overall efficiency calculated from the optimum absorber temperatures of both limiting conditions are shown in Fig. 9. It can be seen from this figure that the concentrator efficiency increases with increasing concentrated solar intensity and with decreasing cooler temperature. Since the optimum absorber temperature for maximizing overall efficiency in the case of maximum possible power output is higher than that of the maximum possible efficiency, then, the concentrator efficiency in the case of maximum possible power output is lower than in the case of

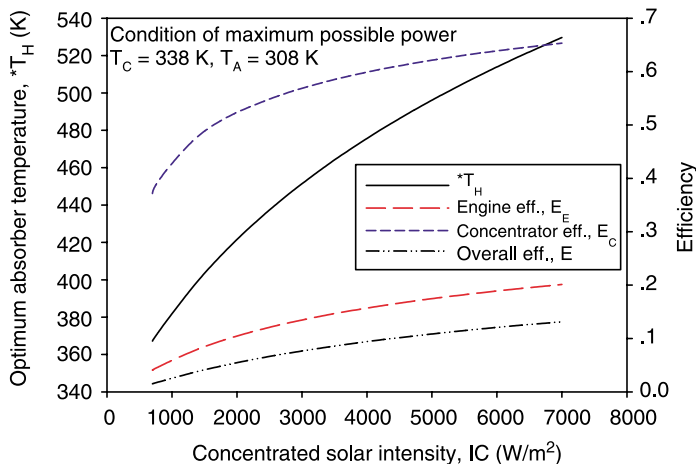


Fig. 8. Optimum absorber temperature and efficiencies against concentrated solar intensity in case of maximum possible power output.

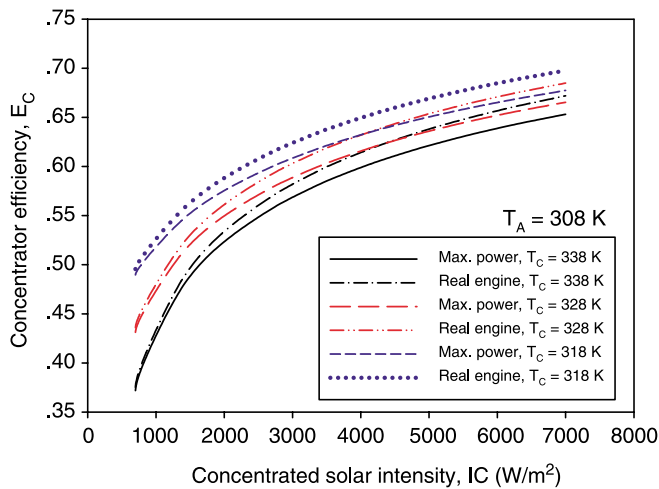


Fig. 9. Concentrator efficiency against concentrated solar intensity, comparison between a case of maximum possible power output and real engine.

maximum possible efficiency. The concentrator efficiency in the case of maximum possible efficiency is used to represent the collector efficiency of the real engine.

Fig. 10 showed relationships between engine efficiency at maximum overall efficiency and concentrated solar intensity in a condition of maximum possible power. It is evidenced that, for all curves, the engine efficiency increases with increasing concentrated solar intensity. The higher engine efficiency can be obtained by using lower cooler temperature.

To compare the condition of maximum possible power with a real engine, the relationships between engine efficiency at maximum overall efficiency and concentrated

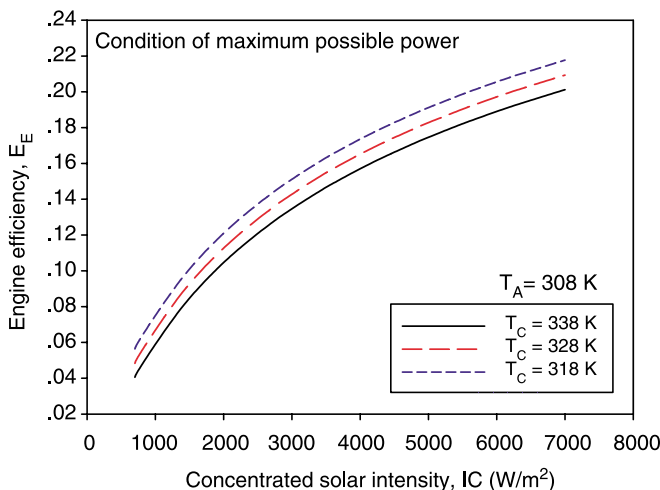


Fig. 10. Engine efficiency against concentrated solar intensity in case of maximum possible power output.

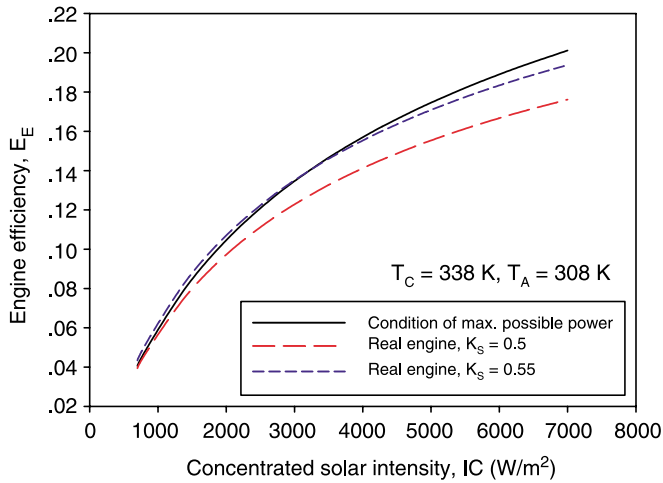


Fig. 11. Engine efficiency against concentrated solar intensity, comparison between a case of maximum possible power output and real engines of $K_S = 0.5$ and 0.55 .

solar intensity in a condition of maximum possible power and real engines of $K_S = 0.5$ and 0.55 are shown in Fig. 11. It can be seen that the engine efficiency in the case of maximum possible power output is very closed to that of the real engine of $K_S = 0.55$.

The relationships between the maximum overall efficiency and concentrated solar intensity are shown in Fig. 12. For all curves, the maximum overall efficiency increases with an increasing concentrated solar intensity. The overall efficiency also increases with decreasing cooler temperature.

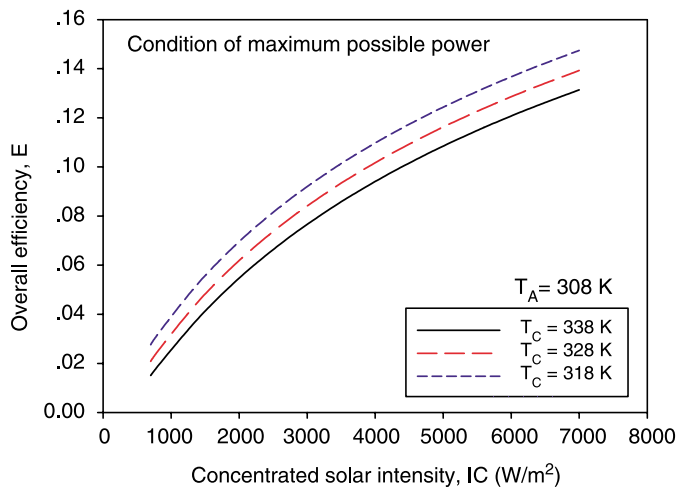


Fig. 12. Overall efficiency against concentrated solar intensity in case of maximum possible power output.

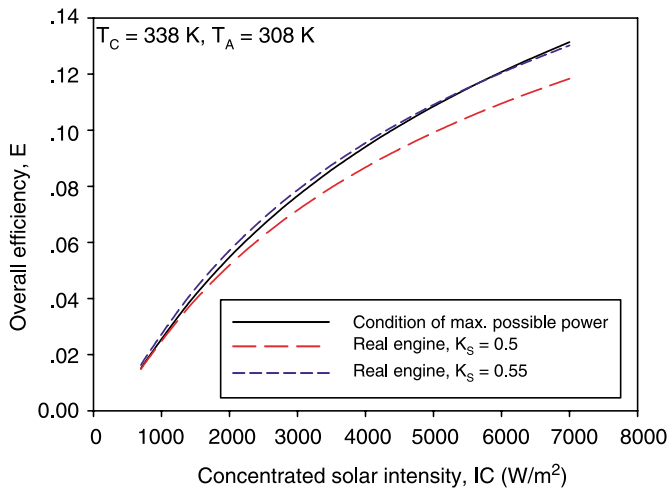


Fig. 13. Overall efficiency against concentrated solar intensity, comparison between a case of maximum possible power output and real engines of $K_S=0.5$ and 0.55 .

To compare the condition of maximum possible power with a real engine, the maximum overall efficiency in the condition of maximum possible power and real engines of $K_S=0.5$ and 0.55 are plotted against the concentrated solar intensity as shown in Fig. 13. It can be noted that the curve of the condition of maximum power is much closer to that of the real engine of $K_S=0.55$.

4. Conclusions

This work provides a theoretical investigation on the optimum absorber temperature of a once-reflecting full conical solar concentrator for maximizing overall efficiency of a solar-powered LTD Stirling engine. A mathematical model for the overall efficiency of a Stirling engine powered by an ideal cone in the general case of absorber heat loss is developed. Two limits of maximum possible engine efficiency and maximum possible engine power output are studied. The optimum absorber temperature for maximum overall efficiency for both limiting conditions is determined. The maximum overall efficiency in the condition of maximum possible power output is compared to that of a real engine. From this study, the following conclusions can be drawn:

1. The results indicate that the optimum absorber temperatures calculated from the condition of maximum possible efficiency and the condition of maximum possible power output are not significantly different.
2. The optimum absorber temperature will increase with increasing concentrated solar intensity and will decrease with decreasing cooler temperature.
3. The maximum overall efficiency will increased with increasing concentrated solar intensity and decreasing cooler temperature.

4. For a given concentrated solar intensity, the maximum overall efficiency calculated from the condition of maximum possible power output is very close to that of the real engine of $K_S=0.55$.

Acknowledgements

The authors would like to express their appreciation to the Joint Graduate School of Energy and Environment (JGSEE) and the Thailand Research Fund (TRF) for providing financial support for this study.

References

- [1] Kongtragool B, Wongwises S. A review of solar powered Stirling engines and low temperature differential Stirling engines. *Renew Sustain Energy Rev* 2003;7:131–54.
- [2] Kongtragool B, Wongwises S. Theoretical investigation on Beale number for low temperature differential Stirling engines. *Proceedings of the second international conference on heat transfer, fluid mechanics, and thermodynamics 2003* [Paper no. KB2, Victoria Falls, Zambia].
- [3] Kongtragool B, Wongwises S. Investigation on power output of the gamma-configuration low temperature differential Stirling engines. *Renew Energy* 2005;30:465–76.
- [4] Howell JR, Bannerot RB. Optimum solar collector operation for maximizing cycle work output. *Solar Energy* 1977;19:149–53.
- [5] Gordon JM. On optimized solar-driven heat engines. *Solar Energy* 1988;40:457–61.
- [6] Eldighidy SM. Optimum outlet temperature of solar collector for maximum work output for an Otto air-standard cycle with ideal regeneration. *Solar Energy* 1993;51:175–82.
- [7] Chen L, Sun F, Wu C. Optimum collector temperature for solar heat engine. *Int J Ambient Energy* 1996; 17(2):73–8.
- [8] Brinkworth BJ. *Solar energy for man*. Compton, Chamberlayne: The Compton Press; 1974 p. 75–6, see also p. 131, 134, 136, 137.
- [9] Senft JR. *Ringbom Stirling engines*. New York: Oxford University Press; 1993. p. 129–31.
- [10] Kreith F, Kreider JF. *Principles of solar engineering*. New York: McGraw-Hill; 1978 p. 530, see also p. 531 and 534.
- [11] Meinel AB, Meinel MP. *Applied solar energy: an introduction*. Reading: Addison-Wesley; 1976 p. 232.
- [12] Walpita SH. Development of the solar receiver for a small Stirling engine, Special study project report no. ET-83-1. Bangkok: Asian Institute of Technology; 1983 p. 33, see also p. 35.
- [13] Daniels F. *Direct use of the sun's energy*. New Haven: Yale University Press; 1964. p. 260.
- [14] Curzon FL, Ahlborn B. Efficiency of a Carnot engine at maximum power. *Am J Phys* 1975;43(1):22–4.
- [15] Reader GT, Hooper C. *Stirling engines*. London: Cambridge University Press; 1983 p. 247, see also p. 248.
- [16] Martini WR. *Stirling engine design manual* 1983 [NASA CR-135382] p. 101.
- [17] Walker G. *Stirling engines*. Oxford: Clarendon Press; 1980. p. 76.